Mechanisms Design for the S5P-TROPOMI Instrument

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Abstract

TNO developed two mechanisms as a module of the Sentinel 5 Precursor (S5P) TROPOspheric Monitoring Instrument (TROPOMI). The TROPOMI is an advanced absorption spectrometer for Earth observation, developed in The Netherlands under contract to NSO and ESA for the ESA Copernicus Space Component Programme. The TROPOMI will allow continuation of the 15-year satellite data sets started with GOME, OMI and SCIAMACHY. This paper will address all design aspects of the mechanism starting with design considerations and ending with the results of a successful qualification campaign. The design challenges of these mechanisms are: a tight design space, the overall mass, the bearing loads due to both a relatively high carousel mass and vibration launch load (~30g quasi static loading which drives the bearing loads), the relatively high required angular reproducibility, and the thermal constraints (operational temperature range).

Introduction

TNO has extensive heritage in designing mechanisms for optical space instruments. In this particular case it concerns the two mechanisms within the instrument needed to fulfill different functionality. One of the mechanisms, the Folding Mirror Mechanism (FMM), requires high positional accuracy and reproducibility on one angular position for the mirror, whereas the second mechanism, the Diffuser Mechanism (DIFM), has to provide six distinctive angular positions for the diffuser carousel with sufficient accuracy and reproducibility. The goal has been to use the same bearing and motor type for the two mechanisms. For reliability reasons a redundant winding motor concept is chosen as baseline for both mechanisms.

Functional description of the Calibration Unit (CU)

The two mechanisms of the Calibration Unit (CU) cooperate to create several optical paths enabling calibration and nadir view. Figure 1 shows the different rotational positions of both mechanisms in section view. The Folding Mirror Mechanism (the smaller mechanism at the right top in the figure) only has one relevant position. It should be able to rotate to enable the slit in the correct position.

The Diffuser Mechanism (DifM) has to be able to rotate the optics in six different positions:

- Solar calibration: this is done in two positions, mutually rotated with 180°
- White Light Source Calibration (90° rotated with respect to the solar calibration)
- Laser diode calibration (180° rotated with respect to the White Light Source calibration position)
- The LED calibration position is rotated with respect to the solar calibration with 45°)

The FMM has two positions:

- Reflecting light of any of the six positions of the DifM
- Passing through Earth radiance

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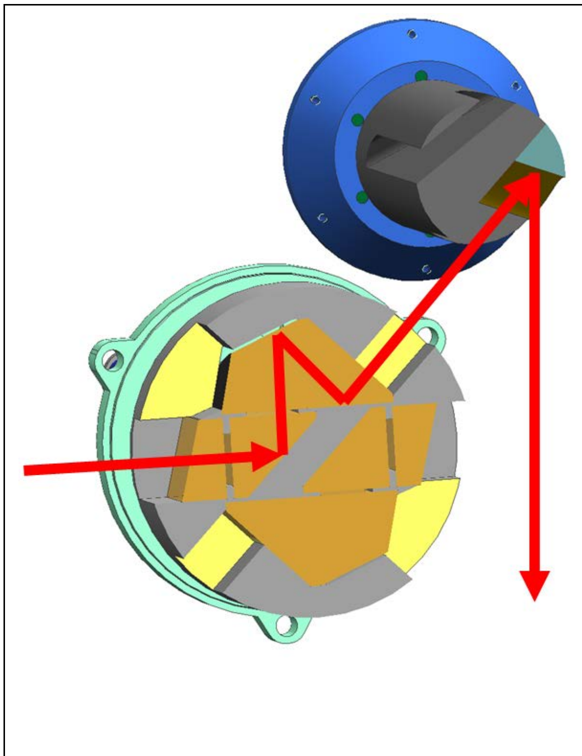
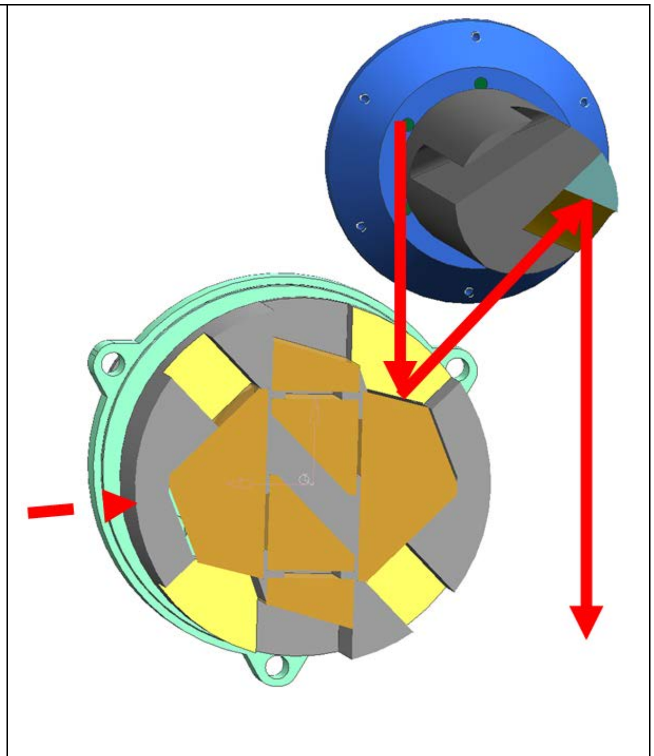
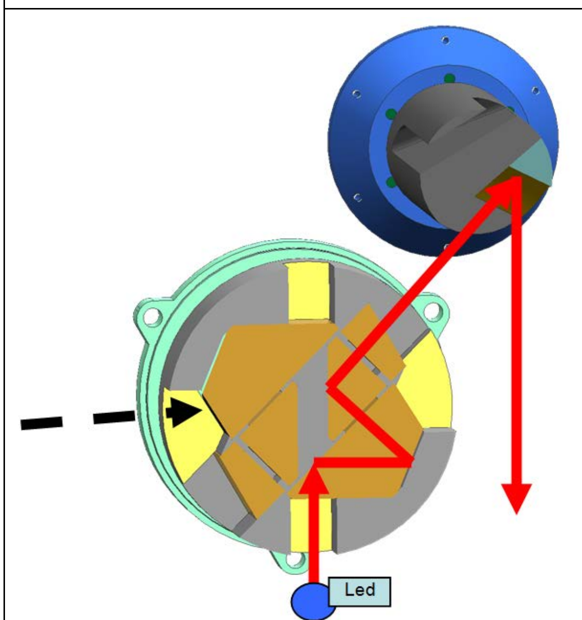
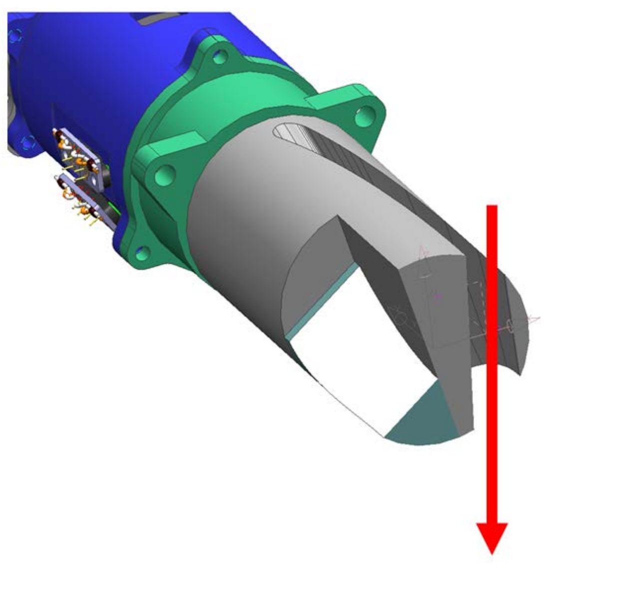
	
Solar Calibration (2x180°)	White Light Source Calibration and (180° rotated) Laser diode calibration
	
LED Calibration	2nd rotational position of the FMM. This position passes through the Earth radiance

Figure 1. The different rotational positions of both mechanisms in section view. The arrows indicate the light paths through the mechanisms

An overview of the driving requirements for both mechanisms is shown in Table 1.

Table 1. Overview of the driving requirements

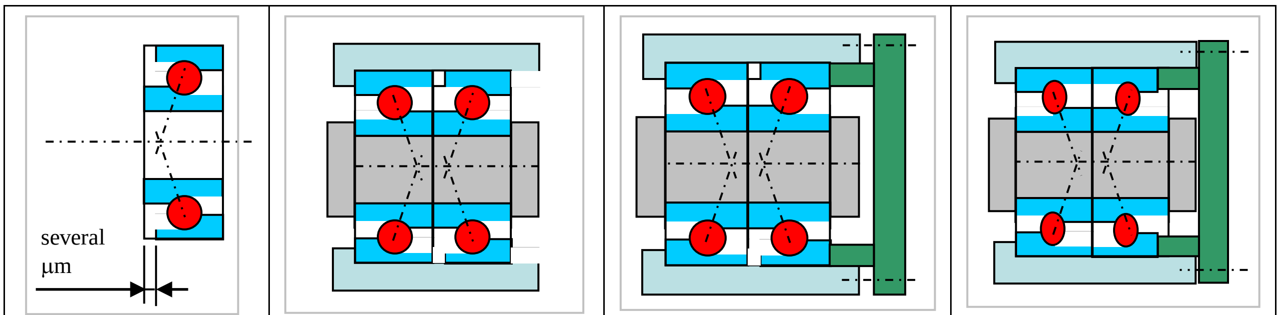
Requirement	Value	Remarks
Reproducibility DifM	RY ± 0.42°	The diffuser carousel shall have a reproducibility of 0.42 degree (zero to peak = 2σ)
Reproducibility FMM	RY ± 0.1°	The folding mirror shall have a reproducibility of 0.1 degree (zero to peak = 2σ).
Testing	-	The folding mirror and the diffuser mechanisms shall be designed to allow complete testing in restricted cleanliness and environmental conditions on ground individually and at spacecraft level.
Number of full rotations in space	30000	The folding mirror and the diffuser mechanisms shall be able to operate for 30000 cycles.
Design survival temperature	-50°C to +45°C	incl. qualification margin
Design operational temperature range	+7°C to +33°C	incl. qualification margin
Rotating Mass (carousel)	3.2 kg	incl. 20% design contingency. Applies to the DifM (worst case)
Eigenfrequency	> 500 Hz	First eigenfrequency of the mechanisms. It concerns here a design goal
Positions	DifM: 6 positions (4x90°, 2x45°) FMM: 2 positions	
Oscillation stroke and frequency	± 5 deg @ 0.5Hz	During a monthly calibration, 3 periods of 10 minutes of oscillations are required
Rotational Speed	90° in 10 s	
Position accuracy	RY ± 0.5 deg	A worst case estimation
Parasitic translations over stroke	ΔX, ΔY, ΔZ< 0.1 mm	A worst case estimation
Parasitic rotations over stroke	ΔRX, ΔRZ< 0.1 deg	A worst case estimation
Number of full rotations for on ground testing	3000	
QS Design Load	30 g	

Bearing Design Considerations

Due to the need for relatively large rotations in the mechanisms, the application of bearings is inevitable. This automatically introduces (dominant) tribology in the system, which together with the application in vacuum, is considered as a design driver. For that reason, the design of the mechanism(s) starts with the design of the bearings.

Initially, a design with two angular contact bearings (in combination with a soft preload) would suffice. This provides a relative low complexity design. One end remains fixed whereas the other end is flexible or able to slide in the axial direction (this is in order to minimize the disturbance forces due to thermo-mechanical loads). However, in order to guarantee surface contact in the bearings (during launch loads), the preload needs to be higher than the forces induced by launch. This yields relatively high required preloads. For the mechanisms as described in this article this would yield negative motorization margins. Additionally, the bearing gapping would become unacceptably high. Both the topics motorization margin and the bearing gapping are treated later on in this article.

A solution to the problem of the high preload and large amount of gapping is by applying duplex bearings. A duplex bearing has the advantage that it can be loaded to high loads in both directions while maintaining a relative small preload. A duplex bearing is a bearing set which is prepared to have a dedicated preload, once clamped together. The principle (as shown in Figure 2) is that the width of all the four rings (the inner and outer bearing rings of both bearing) is measured. Next, it is determined by the bearing supplier how much is required to polish from the outer ring of one of the two bearings. Typically this is in the order of microns with a preload of tens of newtons. Next, the bearings are assembled and compressed together in such a way that the outer rings are in contact with each other. The step-by-step procedure is illustrated in Figure 2. The drawback of applying a duplex bearing is the over-constrained system that it results in: the shaft is fixed in four single bearings (two too many). This will be a point of attention when designing the supporting mechanics.



Several µm is polished from the outer ring of 1 of the 2 single bearings	The inner shaft and the outer housing is assembled	A flange is used to hard preload the bearings	The duplex bearing is preloaded

Figure 2. The principle of a duplex bearing

The bearing properties are summarized in Table 2. These parameters have been established with the aid of the bearing supplier ADR [1] in cooperation with ESTL [2].

Table 2. Bearing properties

Parameter description	Value
Bearing material	Stainless steel 440C
Yields strength bearing material	4000 MPa
Ball diameter	4.762 mm
Loading Configuration	Face-to-face (FF) or X-configuration
Pitch diameter	~ 34 mm
Contact angle	30 ± 2 deg
Ball complement	17 [-]
Width (single bearing)	9 mm
Width (duplex bearing)	18 mm
Compliance (1/Stiffness)	0 m/N (‘hard’ preloading). This is the setting in the analysis software CABARET. For duplex bearings, the axial stiffness of the bearing determines the compliance of the ‘preload spring’
Preload	50 ± 10N
Bearing Center Plane spacing	9 mm
Shaft O.D.	25 mm
Housing I.D.	42 mm
Coefficient of friction (MoS_2 coating, applied on both the races and the balls	0.05…0.08

Motor Choice

Each of the mechanisms is driven with a stepper motor of the same type, the Phytron [5] VSS 52.200. The motor is procured as a complete unit, including internal bearings. Table 3 shows the main properties. The redundant motor windings are wound around separate poles. This avoids failure propagation in the windings. The eight poles are clearly seen in Figure 3 (right) and the picture also shows that each coil is wound around separate poles. A photograph of the complete motor is shown on the left in the figure.

Figure 3. Photograph of the Phytron stepper motor (left) and a photograph of a similar motor (right) provided by Phytron

Table 3. Motor properties

Parameter description	Value	Remarks
Motor Type	VSS 52.200.1.2	
Supplier	Phytron	
Number of teeth	50	
Number of phases	4	With the 50 teeth this results in a full step resolution of 1.8 ° (200 steps/rev)
Number of poles	8	
Windings	"cold" redundant	
Holding Torque	> 0.320 [Nm]	
Running Torque	> 0.220 [Nm]	

Conceptual Design Description

The design of both mechanisms is optimized to the ability to survive launch loads and having a low friction torque and therefore a high torque margin. Furthermore, the mechanisms are designed as compact as possible to limit the overall mass and volume. The principle of the design baseline is identical for both mechanisms (see Figure 4) and can be roughly divided in the following features:

- A stepper motor drives the carousel of the mechanisms
- The carousel is symmetrically supported by bearings (one duplex at each side)
- The motor and the carousel are coupled through a flexible coupling. This coupling is torsionally stiff, but compliant in all other directions
- The carousel consists of:
    - For the DifM: A set of 6 Fused Silica components (diffusers)
    - For the FMM: A slit and a folding mirror

The left duplex fixes 5 DoF (only Rz is free), but due to the hinge the Rx and Ry are released. The X and Y direction are still constrained since the rotation point of the hinge shaft is placed symmetrically within the duplex bearing. The Z direction is also fixed by the left duplex bearing. The right duplex bearing is fixed in X, Y by the membrane. The Z direction is left free which allows for thermal expansion differences between the carousel (Invar) and the fixed world (aluminium). Hereby, the preload of ~50 N remains approximately the same at large temperature excursions. The Rx and Ry of the right duplex are left free due to the membrane. Since both the carousel and the motor shaft are fixed in all except the rotation, a flexible coupling is needed which only transfers the Rz in a stiff manner.

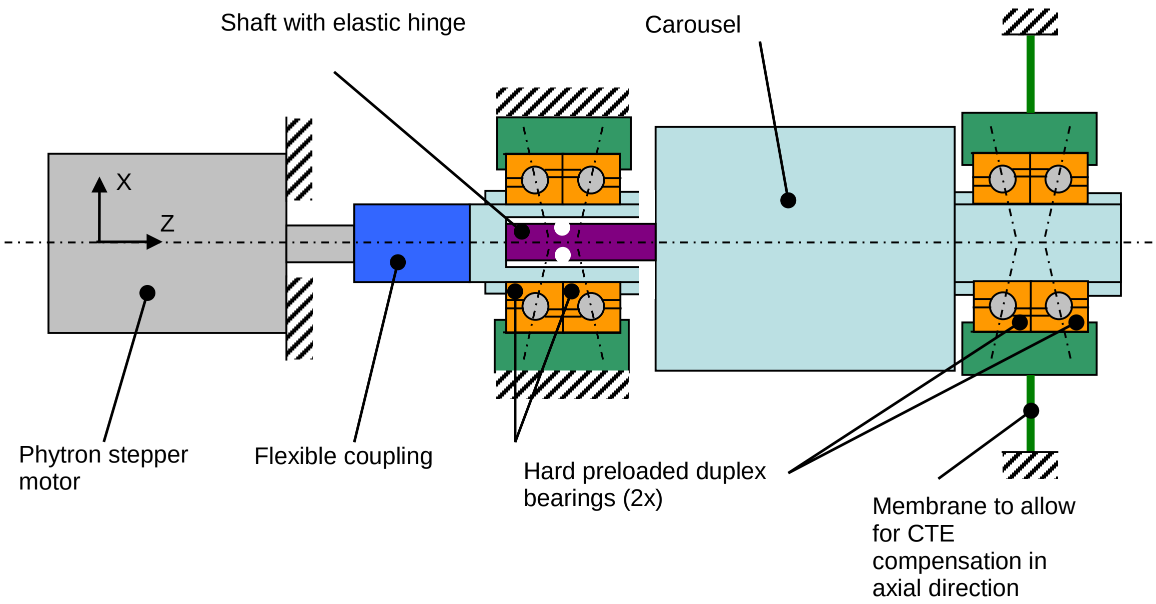


Figure 4. Schematic overview of the concept design of the mechanism

Detailed Design Features

<u>Hinge shaft</u>

The elastic hinge is introduced in the design in order to allow for a kinematic design. Therefore, the hinge will ensure that the design is predictable and reproducible even with the presence of tolerance chains within the rotor and relative to the CU housing. The predictability is also guaranteed during the thermal survival and operational range. The following aspects are of importance with respect to the hinge:

- Stresses due to launch loads (30g Quasi-Static load in axial and radial direction)
- Bending effects (stiffness and stresses)
- Stiffness in axial and radial direction (driven by the required > 500 Hz)

Figure 5 shows a side view with the two main parameters: the diameter of the hinge (D = 6mm) and the width (h = 4 mm). The shaft is manufactured from Ti6Al4V, which is a close thermal match (CTE match) with the 440C bearings.

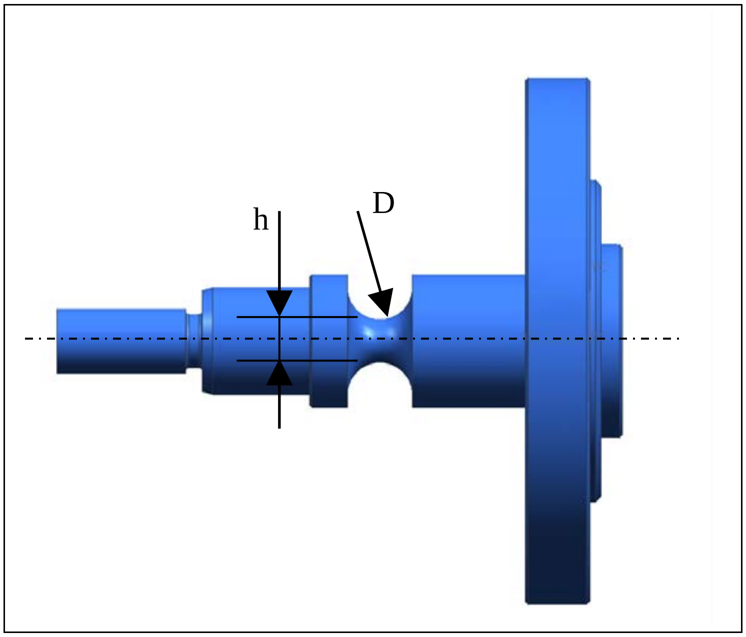


Figure 5. A side view of the elastic hinge with the two main parameters: the diameter of the hinge (D) and the width (h)

<u>Flexible coupling</u>

The flexible coupling transfers the motor torque to the carousel. Note that this coupling is designed to be torsionally stiff (around the center line) and compliant in the 5 other DOFs. This coupling is needed to enable a kinematic and therefore predictable and reproducible design. Note that the motor is equipped with its own bearings and that the motor housing is rigidly attached to the CU housing. The motor shaft has only one DOF free and that is the rotation around the center line. Also the carousel has only one DOF free, i.e., rotation around the center line. It is therefore essential that the coupling does not add additional stiffness in the remaining 5 DOFs which would result in over-constraining the system. The design of the coupling is illustrated in Figure 6.

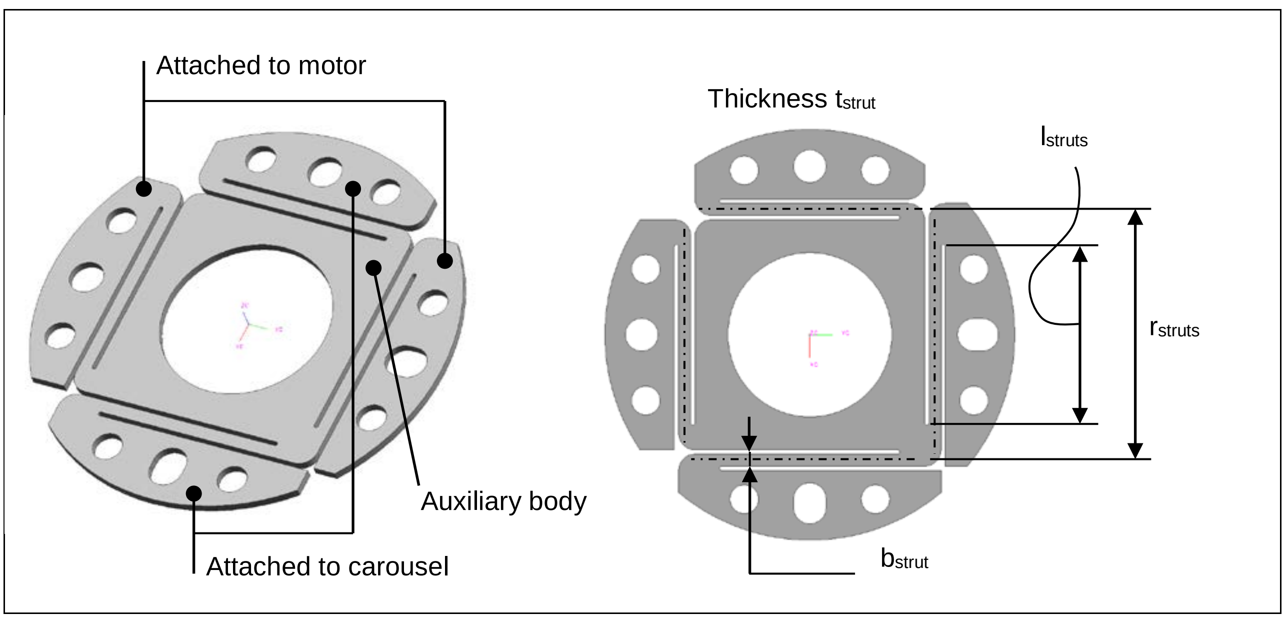


Figure 6. Flexible coupling used to couple the drive shaft of the motor to the rotor

The coupling is a monolithic part (Ti6Al4V), manufactured by wire EDM from one single piece of plate. The result is two pairs of flanges, mutually coupled by four struts. One flange pair is rigidly mounted to the motor side. Via one pair of parallel struts, the torque is transmitted through the auxiliary body. Next, through the second pair of parallel struts the torque is transmitted to the carousel. The principle of the coupling is also known as an 'Oldham' coupling in literature.

The only drawback of this coupling is the fact that the auxiliary body is not fixed in axial direction. This stiffness is provided by the lateral stiffness of the struts which is designed to be as low as possible. Special attention is given to keep the first eigenfrequency of this auxiliary body sufficiently high.

<u>Membrane</u>
One of the two duplex bearings of each mechanism is supported by a membrane (Figure 7). The reason for this is to allow for CTE differences (axially) between the carousel and the CU housing. A membrane will result in a predictable design. An alternative would have been to use a sliding fit. However, this is less predictable and reproducible than an elastic solution with a membrane. The dimensions of the Ti6Al4V membrane are shown in the figure.

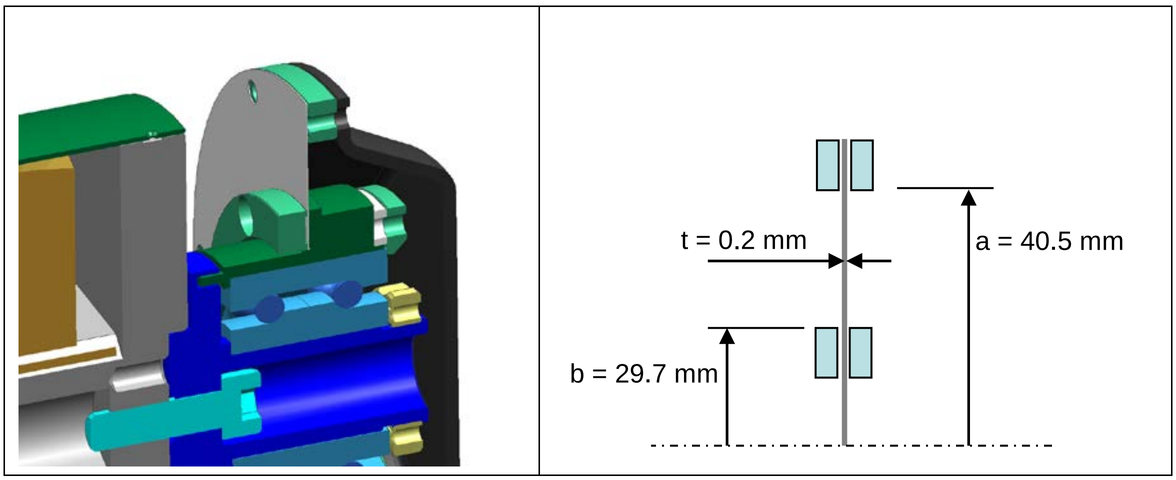


Figure 7. CAD picture (left) and general dimensions of the membrane

<u>End stop</u>
Contrary to the DifM, the FMM is equipped with an end stop. The reason for this is that a higher reproducibility can thereby be achieved for the FMM. Using the detent torque of the stepper motor, a positive torque is exerted against the end stop which enables a stable position with an unpowered motor. The FMM has only two positions: one of the folding mirror (which has to be positioned reproducible) and one for passing through the Earth's radiance (oversize sloth). The end stop is shown in Figure 8.

A cam disc is rigidly attached to the rotating shaft. Additionally, a key or spline is used to prevent mutual rotation. The cam will rest on a Vespel end stop. This end stop, which is clamped to the end stop holder, is adjustable with a shim. This material combination was chosen because:

- Vespel combined with metal avoids cold welding.
- Vespel was used in the Sciamachy and OMI projects, also as an end-stop. Both passed a (comparable) life test.

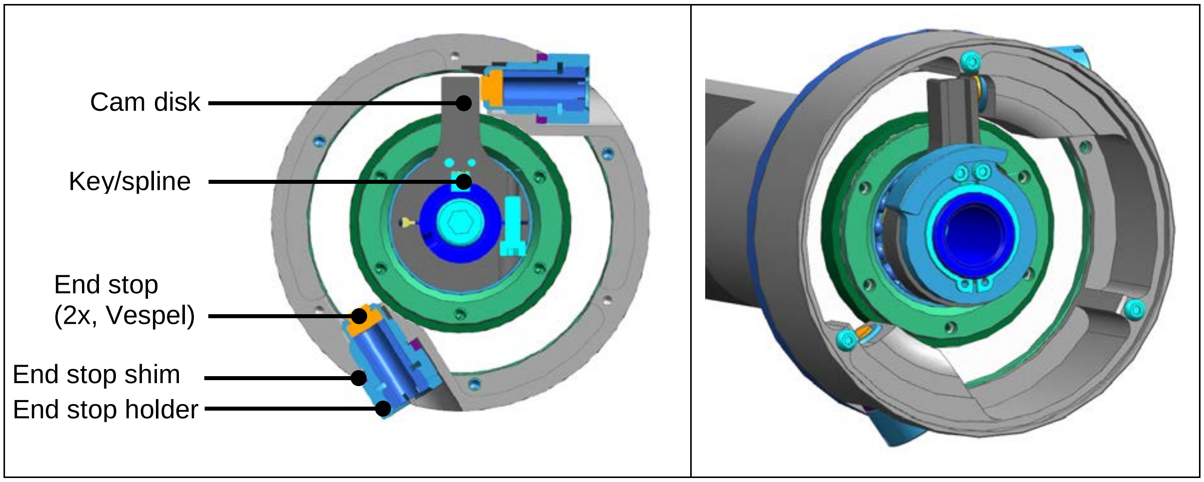


Figure 8. The FMM is equipped with an end stop in order to guaranty an angular reproducibility < 0.1°. A cam disc will rest on an adjustable Vespel end stop. Note that two stops are incorporated.

Bearing analyses

The following analyses are considered to be relevant for assessing the feasibility of the bearing concept:

- Gapping and stress in the bearings due to axial and radial loads (Quasi-Static loads)
- Bearing friction torque due to preload (@ T=20°C)
- Bearing friction torque due to preload (@ worst case operational range)
- Bearing stiffness calculations (radial and axial)

The bearing analyses have been carried with the assistance of the European Space Tribology Laboratory (ESTL) and using the software CABARET. The principle of gapping is explained in Figure 9. Both for radial and axial gapping, an unwanted physical gap will be present. The idea is to minimize this gap. The maximum allowable gapping is set to 20 μm. This is based on the approach followed in the EarthCARE MSI-VNS mission [3] and is based on communication with ESA (European Space Agency) and ESTL. Due to the maturity of the CU design, the design goal for gapping is set to 10 μm.

Figure 10 shows the results for a varying axial load between -2000 to 2000 N. Both the stress and the axial displacements of each of the two bearings are calculated. The axial force for the mechanism is calculated with $F_{ax} = ma = 3.2 \cdot 30 \cdot 9.81 = 941$ N, where the mass *m* of the carousel is 3.2 kg and a 30 g QS load is applied. The bearings start to gap at the point where the Hertzian stress becomes zero. The displacement at this point is 1 μm. The displacement at 941 N is 10 μm. Therefore, the gapping at 941N is 9 um. This is within the design goal of 10 μm.

The maximum stress at 941 N is 1800 MPa. With an allowable stress of 4000 MPa, the Margin of Safety becomes MoS = [(4000 / 1.25) / 1800] -1 = 0.77 which is > 0 and therefore acceptable.

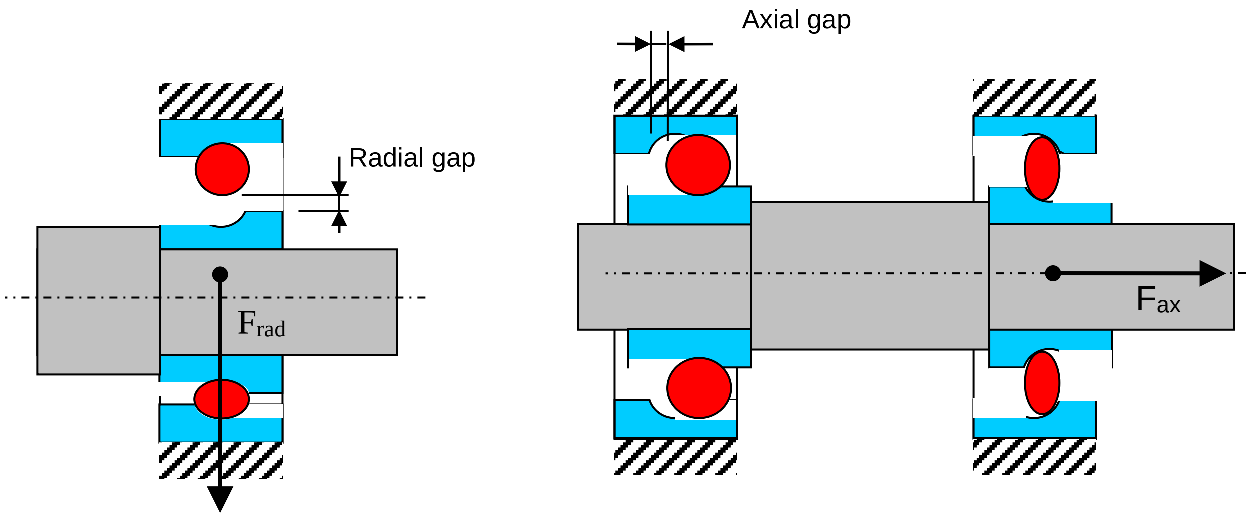


Figure 9. Radial and axial gapping illustrated

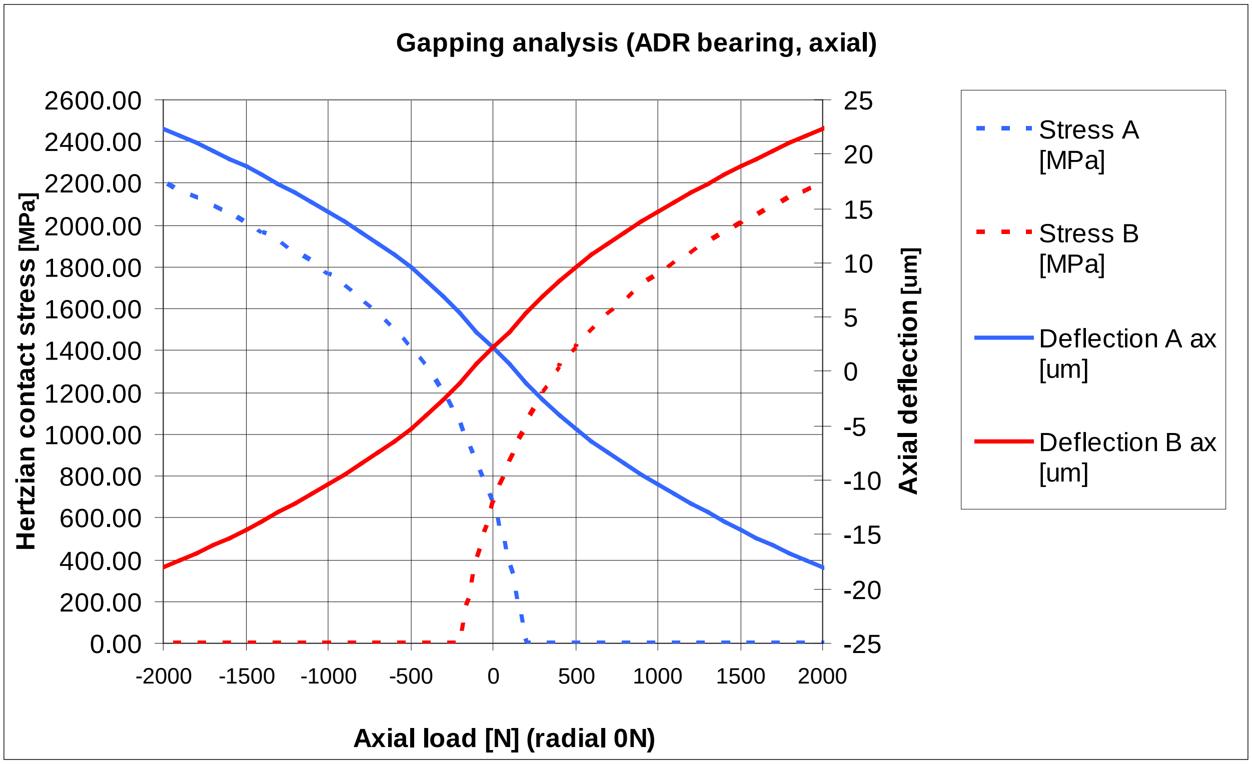


Figure 10. The axial deflection and the Hertzian contact stress in the duplex bearing as a function of the axial load. The preload is set to 50 N.

For two reasons, radial gapping is far less than axial gapping. Firstly, the radial force on the duplex is estimated to be 941/2 = 470 N which is half the axial force. Secondly, due to the contact angle, the radial loading is in a much more favorable direction. At 470 N, the radial gap is ~3 μm which is smaller than the axial gap and thereby also below the design goal of 10 μm. This shows that the duplex bearing is considered to be very suited for radial loads.

Motorization Margin Analyses

The ability of a drive motor to be able to drive a mechanism can be quantified using the Motorization Margin or Torque Margin (MoS_{TORQUE}). This margin is defined as $MoS_{TORQUE} = [\, T_{MIN,MOTOR} / T_L \,] - 1$. Herein, $T_{MIN,MOTOR}$ is the minimal motor torque available and T_L is torque of the load. The latter is defined as $T_L = 2*(FoS_{FRICTION} * T_{FRICTION})$. Herein, $FoS_{FRICTION} = 3$ which is the Factor of Safety on friction and $T_{FRICTION}$ is the torque in bearings. According to ECSS [4], inertia and spring effects should also be taken into account, but for these mechanisms these budgets are considered negligible. The torque margin is calculated for the following load cases:

- At 20°C
- At worst case operational range

The following inputs are elaborated on. The friction torque is calculated with CABARET as described in the previous section. It concerns here a mean friction torque with a worst case coefficient of friction. Next, in order to obtain the maximum (0-peak) friction torque, the mean friction torque is multiplied by a factor of three. This factor is based on empirical data as deduced by ESTL. Next, margins according to [4] are incorporated in the calculations. The resulting margins for both cases are 3.1 and 2.6 respectively, which are both > 0. From this it is concluded that the motor is suited for the mechanisms.

Life Test Model Qualification

The Tropomi Life Test Model Calibration Unit has successfully passed the life test program, consisting of:

- Vibration testing (see Figure 11) with a 9.71 g_{rms} input spectrum, resulting in 28.6 g_{rms} on the carousel
- Thermal Vacuum Testing (from -50°C to +45°C) as shown in Figure 12
- On-ground cycles and in-orbit cycles up to 72000 cycles

No change in performance between start and end of the life test program, determined by:

- Pre and Post Vibration resonance search
- Minimum start current (torque margin) during the life test program
- Destructive Physical Analysis

References

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3. Tabak, E., de Goeij, B., van Riel, L., Meijer, E., van der Knaap, F., Doornink, J., de Graaf, H. "Design, building and testing of a Sun Calibration Mechanism for the MSI-VNS Instrument on EarthCARE." *ESMATS 2013*.

4. ESA-ESTEC, ECSS-E-ST-33-01C Mechanisms, Requirements & Standards Division Noordwijk, The Netherlands, *6th March, 2009*

5. Phytron, www.phytron-elektronik.de

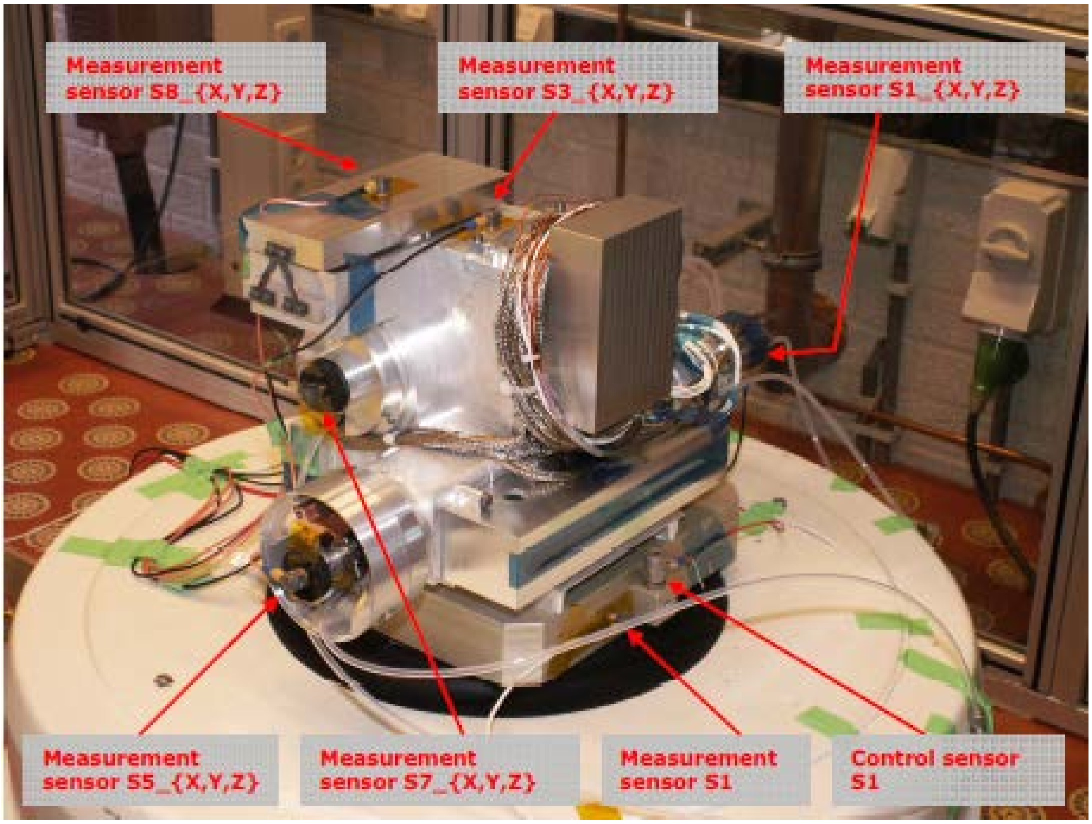

Figure 11. Vibration testing of the CU with a 9.71 g_{rms} input spectrum, resulting in 28.6 g_{rms} on the carousel

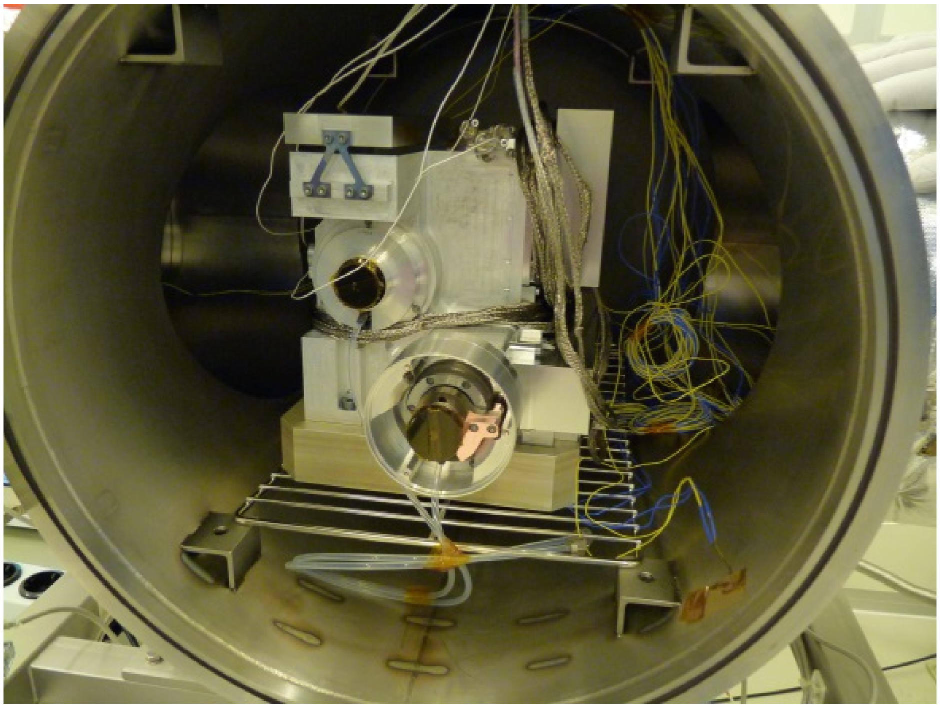

Figure 12. Thermal Vacuum Testing of the CU